

Modeling of Active and Passive Damping Patches with Application to a Transmission Casing Cover

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ABSTRACT

Combined active and passive damping is a recent trend that can be an effective solution to challenging NVH problems, especially for lightweight vehicle components that demand advanced noise and vibration treatments. Compact patches are of particular interest due to their small size and cost, however, improved modeling techniques are needed at the design stage for such methods. This paper presents a refined modeling procedure for side-by-side active and passive damping patches applied to thin, plate-like, powertrain casing structures. As an example, a plate with fixed boundaries is modeled as this is representative of real-life transmission covers which often require damping treatments. The proposed model is then utilized to examine several cases of active and passive patch location, and vibration reduction is determined in terms of insertion loss for each case. Results are compared to an experiment with an actual transmission casing for validation, using piezoelectric active patches and constrained-layer passive patches with a viscoelastic core. Conclusions are drawn about patch size and location in terms of NVH reduction capability, and guidelines are suggested for the dynamic design process.

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INTRODUCTION

Current lightweighting trends in vehicle design improve fuel economy, but as a result vehicle components can become more susceptible to unwanted vibrations and structure-borne noise. Additionally, modern lightweight materials (such as aluminum) often have less inherent structural damping, resulting in increased vibration amplitudes. One example is a transmission casing cover, a vehicle component that is often excited by structure-borne vibration in the drivetrain system and that is frequently an efficient radiator of sound [1]. Furthermore, the geometry of such a cover may include irregularities that increase its radiation efficiency. Gear mesh frequencies common in the mid-frequency range (200-1500 Hz) are responsible for exciting modes of these covers. Damping solutions to reduce noise radiated by large-amplitude vibration are needed that are both lightweight-keeping with the lightweighting trends discussed-as well as effective in terms of both size and cost.

Passive constrained layer damping patches have been shown to be an effective damping solution for these types of problems [1-2]. Such patches typically use a viscoelastic core with a large material loss factor (tuned to a given temperature range) and dissipate vibrational energy via shear deformation. Certain active patches (in particular piezoelectrics) have also been studied for vibration reduction applications both passively, in a shunted-circuit configuration [3], as well as actively, in a destructive interference sense [4]. The latter method introduces vibration to the system via a control signal intended to interfere with unwanted vibration from a disturbance (in

this case, structure-borne vibration from the powertrain). Combined active and passive methods have been proposed previously in the form of "active constrained layer damping" [5] with active materials embedded in traditional passive damping patches. However, little research has been done on concurrent "side-by-side" active and passive damping to find the effect of passive damping patches on control from an active patch. As such, improved modeling methods are needed to determine the potential benefits of such patch configurations, as well as to provide insight to the design process.

While both active and passive patch methods have been applied to a variety of simple structures including classical beams, plates, and shells [3, 4, 5, 6], realistic components are more difficult to model as they often include curvature, features such as fillets and varying thickness, and complex boundaries such as bolted joints. Accordingly, this paper seeks to address this void in terms of application-based modeling with the following objectives. (1) Perform an experiment to determine the dynamic behavior of a transmission casing cover. (2) Propose a design-oriented analytical method for modeling of active and passive patches (in the "side-by-side" configuration) to be implemented on the plate-like cover. (3) Quantify the efficacy of active and passive patches for reducing vibration in terms of modal loss factor and insertion loss, and evaluate other qualitative benefits of combined active and passive methods. (4) Validate the model with an experiment. (5) Propose design guidelines in terms of patch location and placement for given operating conditions.

TRANSMISSION CASING EXPERIMENT

A production transmission casing cover (shown in Figure 1a) is the motivation for this research. However, to simplify the experiment and allow for more tractable analytical models, a modified casing and cover are used (as in Figure 1b). The modified casing, made of aluminum, is significantly stiffened and has significant extra mass added to eliminate any structural casing modes from the frequency range of interest. The modified casing cover has the same shape as the production cover but has no out-of-plane features, resulting in a flat plate, as detailed in Figures 1c vs. 1d. The steel cover is affixed to the casing via 10 bolted connections and has approximate major dimensions of 300 mm $\times$ 180 mm. Two covers are used for the preliminary experimental study, of thickness 1.42 mm and 2.92 mm.

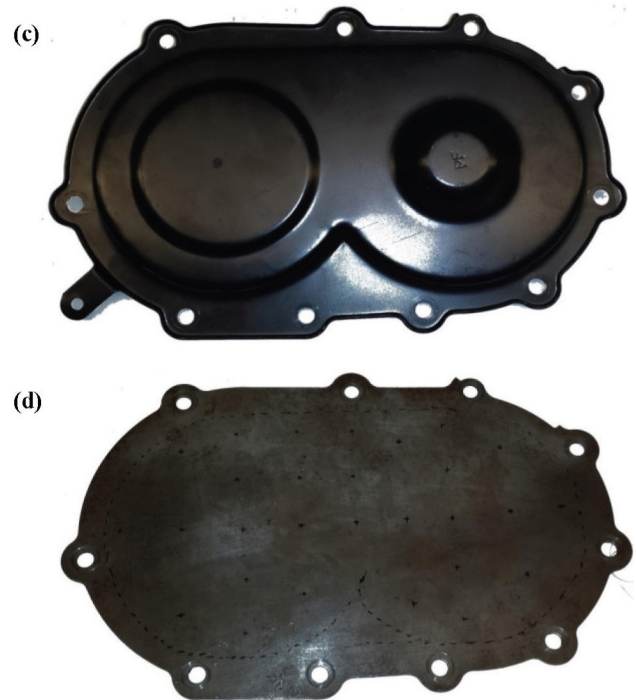
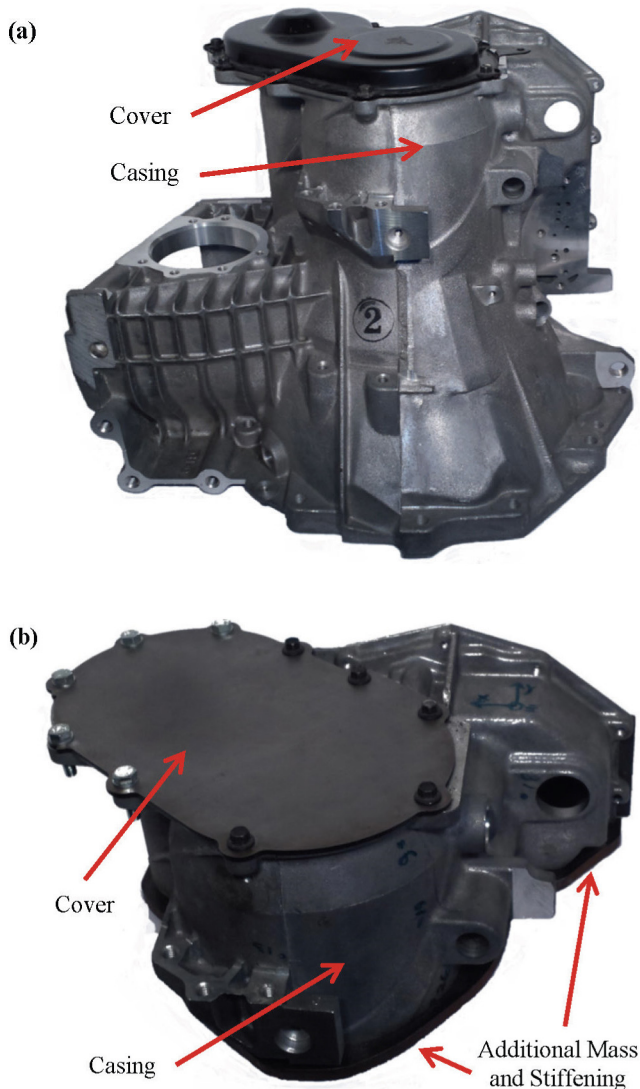


Figure 1. (cont.) Transmission casing and cover: (a) production sample; (b) simplified version for experimental study; (c) detail of production cover; (d) detail of simplified cover

Accelerance measurements, defined as acceleration per unit force, are made of the cover in the *in-situ* condition (bolted to the casing) with an impulse hammer. A schematic of this procedure, as well as force and measurement locations, are shown in Figure 2a.

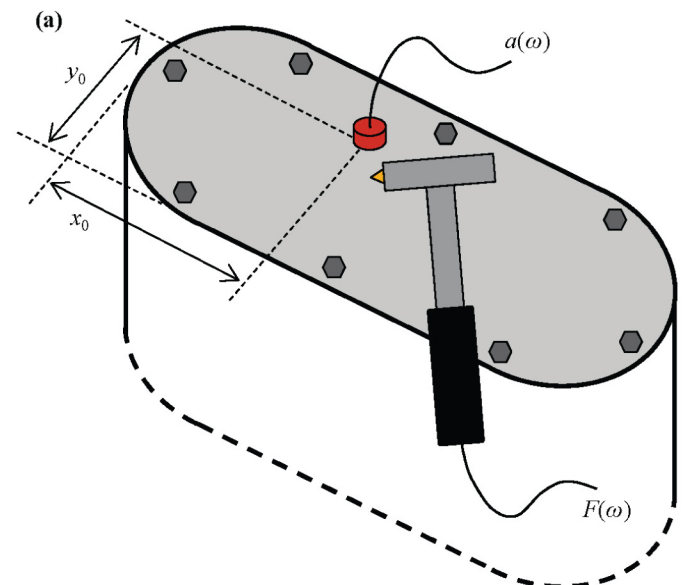
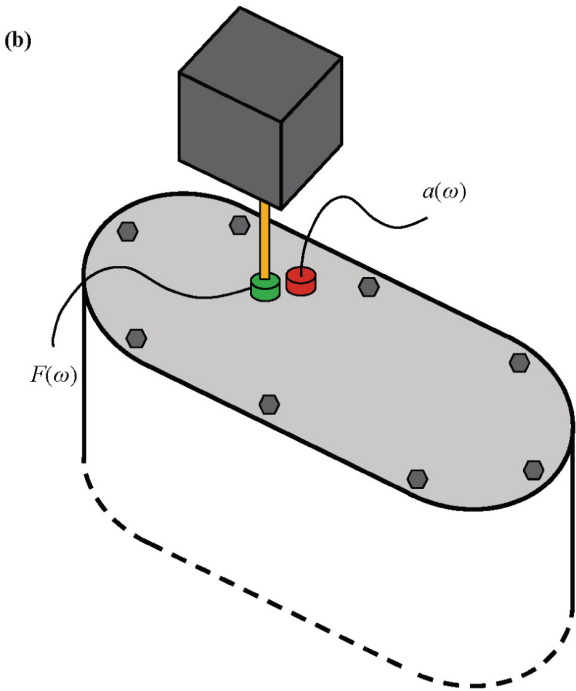


Figure 2. Schematic of experimental procedure on casing cover: (a) impact hammer with driving point location shown $(x_d, y_d) \approx (0.44 L_x, 0.71 L_y)$; (b) shaker input for driving point measurement

Figure 1.



Driving-point and cross-point measurements are made at 28 different locations (using roving-hammer type measurement methods) in order to extract mode shapes. A least-squares parameter estimation technique is used to obtain natural frequencies, modal loss factors, and mode shapes. The accelerance spectra (driving point) for both cover thicknesses are plotted in Figure 3. Due to better experimental results and modal correlation for the thin cover, this case will be used for the remainder of the paper. Furthermore, the second, or (2, 1) mode, will be considered as the “mode of interest” because it is well isolated, has low damping and a well-defined mode shape, and it is near the middle of the targeted frequency range. Natural frequencies for each mode, along with modal damping ratio and modal index are given in Table 1. Modal indices are defined as (m, n) , where m and n are the total number of local extrema in the long (x) and short (y) dimensions, respectively. As an example, the measured flexural displacement of the (2, 1) mode is shown in Figure 4a.

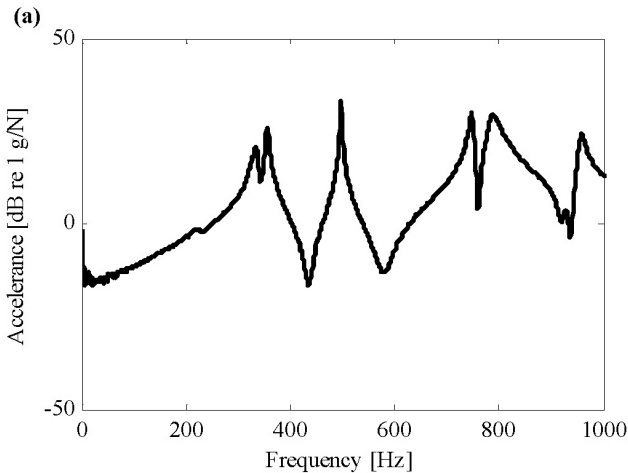


Figure 3.

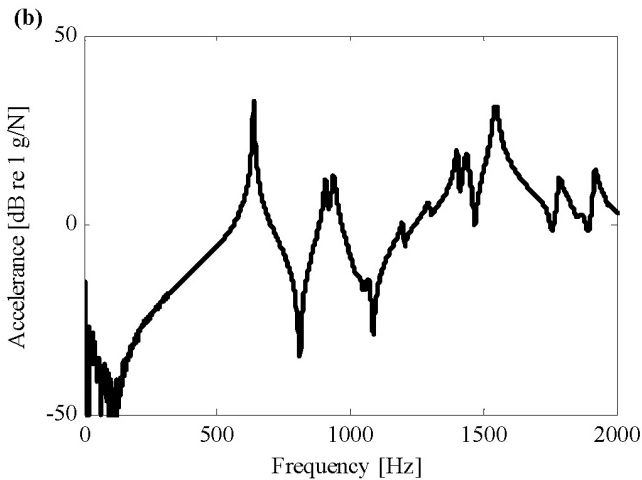


Figure 3. (cont.) Experimental driving-point accelerance magnitude spectra of casing cover: (a) thin cover, thickness = 1.42 mm; (b) thick cover, thickness = 2.92 mm

Table 1. Selected modes of thin casing cover

Mode #	Modal Index (m, n)	Exp. Natural Frequency, f_n [Hz] (Modal Loss Factor, η_n): Realistic Boundaries	FEM Natural Frequency, f_n [Hz]: Fixed Boundaries
1	(1, 1)	355 (0.015)	493
2	(2, 1)	497 (0.002)	587
3	(3, 1)	748 (0.008)	883
4	(1, 2)	787 (0.022)	1255
5	(2, 2)	956 (0.011)	1301

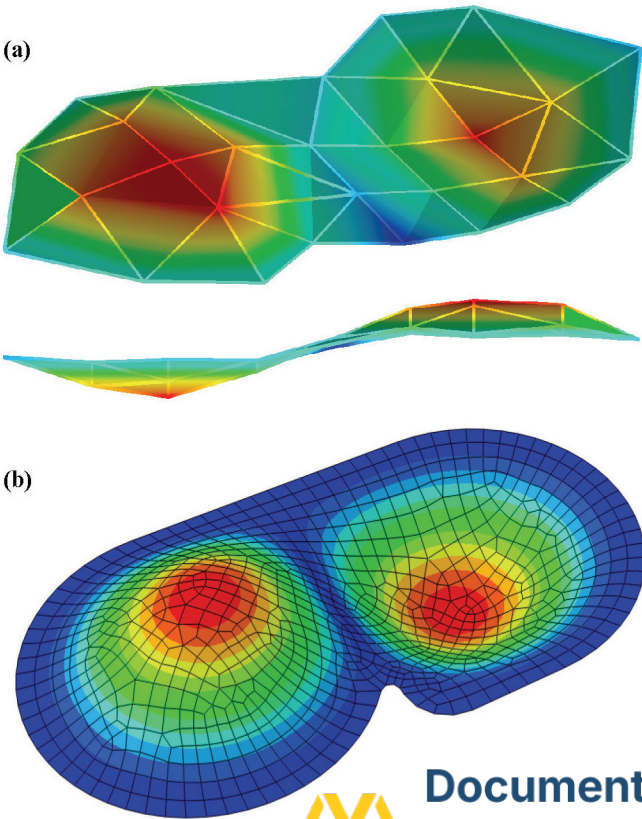


Figure 4. Flexural displacement magnitude at (2, 1) mode: (a) measured, isometric and side views; (b) finite element model